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MEASURED PERFORMANCE OF PUMP IMPELLERS

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Summary

The initial studies conducted in the Hydraulic Machinery Laboratory at the California Institute of Technology on a conventional three-dimensional centrifugal pump impeller, operating free of its case, were reported in November 1949. Since that time detailed experimental work has been conducted on a series of four impellers, all essentially of the same specific speed but each embodying certain design variations. Qualitative studies have been made of the flow patterns at the inlet, in the passages, and at the discharge of each impeller at flow rates from zero capacity to zero head. Using for the most part the same techniques as described in the original report, the head-capacity characteristics of each impeller have been determined. Input power measurements have been made on two of the impellers, thus making it possible to obtain the impeller efficiencies. The results of these studies are presented and analyzed and correlated with the several design variations. The operating characteristics of two of the free impellers are compared with the performance curves obtained from in-case or complete pump tests using the same impellers. Conclusions are reached as to the significance of the volute in determining the over-all operating characteristics of a centrifugal pump.

Introduction

The increasing demand for high-efficiency hydrodynamic machines to meet a broad range of operating conditions had made it desirable to undertake detailed studies of the complete performance of the individual components of such machines. To meet this need a program of basic research on hydrodynamic machines is in progress in the Hydraulic Machinery Laboratory of the California Institute of Technology under the sponsorship of the Office of Naval Research, Fluid Mechanics Branch.

In conducting an investigation of this type new techniques must be developed and old ones revised as each step reveals more of the nature of the flow in an impeller. It follows that conclusions drawn during the early stages of the work are inherently tentative and contingent upon subsequent verification. It is the purpose of this paper to present a current report of the procedures, results, and conclusions.

In the literature there are no reports of studies of the performance of isolated, three-dimensional impellers operating throughout the entire pumping range. References cited in a previous paper (1)^{2/}, describe work which was done using simplified two-dimensional impellers operating over a limited range. The experimental difficulties encountered in operating an impeller free of its case and the complex nature of the flow in three-dimensional impellers have discouraged investigations.

^{2/} Numbers in parentheses refer to bibliography

Laboratory Arrangement

The general laboratory arrangement, including the impeller mounting, has been described elsewhere (1). The flow was delivered to the impeller eye by a nozzle which exactly matched the eye diameter and which was designed to give a uniform velocity profile at that section. The impeller was mounted directly above the nozzle and was rotated about a vertical axis. The impeller discharged radially into a stationary, symmetrical, parallel plate collector which matched the impeller discharge breadth and was fitted at its outer diameter (18-1/2 in.) with two opposed cylindrical weirs. The opening between these weirs was adjustable. Since an impeller so mounted operates without inlet and discharge asymmetries, usually present in normal pumping units, it has been termed a "free impeller".

The Test Impellers

The significant dimensions of the four impellers are shown in Table 1. Two of the impellers, one five and one six vane, were manufactured entirely of lucite and were close approximations of the final Grand Coulee design, which had a specific speed of 102. These impellers had cylindrical discharge vane tips and a relatively small inlet vane angle (2). Unlike the prototype, the spoon or warp in the vane surfaces in the eye was omitted and the leading edges of the vanes were rounded with a 1/16 inch radius. The only difference between Impellers I and II was in the number of vanes.

The other two impellers, one five and one eight vane, were the same metal impellers used in the in-case model studies conducted at the Laboratory during the development of the final Grand Coulee design. The design specific speed of Impeller III was 92, of Impeller IV, 102. The discharge vane tips of these impellers were twisted or skewed and the inlet vane angles were greater than those of the lucite units, the former inlet angles being calculated assuming a small positive prewhirl; whereas, the latter allowed for a negative prerotation at the vane tips at the design point (2). The vane surfaces at inlet were slightly spooned and the inlet edges were filed sharp. These two impellers had different discharge vane angles.

Flow Observation Techniques

Qualitative observations of the flow were made with thread streamers and with fluid particle tracers released in the flow and observed through the medium of high-speed two- and three-dimensional photography. The thread streamers were short in length and loosely fastened to fine wire probes. They were used to determine the approximate flow directions at the discharge of the impellers and in the collector. The tracer technique allowed for more accurate studies to be made not only of the discharge flow pattern but also of the flow at inlet and in the impeller passages. Although the tracer studies were made only on the six-vane transparent impeller, it is assumed that the large scale phenomena to be discussed were present in all the impellers. This assumption seems reasonable since the general character of the discharge flow patterns of all four impellers was found to be much alike.

The two-dimensional motion pictures were taken in the absolute and the relative frame of reference. Pictures taken with a fixed camera mounted above a transparent impeller show the absolute paths of the tracer particles and are extremely useful in studying entrance and discharge flow patterns and the angular momentum history of a particle throughout the impeller. Although possible, it is rather difficult in the absolute field to examine the flow in relation to the rotating impeller vanes. If the impeller is viewed in the relative frame of reference the difficulty is overcome.

To observe the relative flow patterns photographically the camera may be rotated with the impeller or a stationary camera may be used with an auxiliary optical device. This device has been termed in the literature a "rotoscope" (3) and consists of a "dove" or inverting prism located on the axis of the impeller and rotating at one-half of the impeller speed. Fig. 1.

Although obviously advantageous from a mechanical standpoint, there is one disadvantage to the rotoscope technique. A true undistorted image is possible only if parallel light rays pass through the prism, since it is in this case only that all rays travel equal distances within the prism. Hence, if results comparable to those obtainable by the rotating camera technique are desired, the rotoscope must contain not only the prism but also a proper lens arrangement to collimate the entering rays, pass the bundle through the prism, and then reconverge them into the camera objective. Such a lens system must be manufactured to the same precision as the specific camera lens to be used; a requirement which is difficult and costly to meet. Fortunately, a compromise can be made which results in pictures satisfactory for most qualitative work. If a relatively long focal length lens is used, such that for the coverage desired the total entrant cone angle is of the order of 8° or less, the linear distortion will be about 1 or 2 per cent. As the angle increases the distortion increases rapidly and soon becomes objectionable. A 63 mm. lens, $f/2.7$, covering a standard 16 mm. motion picture frame was found satisfactory.

As a result of purely practical considerations, if a collimating system is not used, the prism may reduce the amount of light striking the film. This arises from the fact that it is usually either impractical or impossible to obtain a prism of such proportions as to pass all the entrant rays that would normally be gathered by the camera objective. For crown glass with the refractive index $\mu = 1.52$ and 45° entrant and exit face angles, a rough rule for the prism height-to-length ratio is one to four. For the 63 mm. lens mentioned, which had a lens diameter of 1 inch, a prism approximately 2 x 2 x 8 inches would be required to operate the lens at its largest rated aperture. Prisms up to 2 1/2 inches long are readily available commercially, but larger units must, as a rule, be custom made. If the ray angle is greater than approximately 8° a moment's reflection will show that generally it is impossible to avoid this stopping down. In high speed underwater photography where lighting at its best is often inadequate, this reduction of light caused by the prism is significant. Satisfactory black and white exposures were made with a 0.6 x 2.4 inches prism. The total continuous light power input of 12 kw was concentrated about the impeller at an average distance of 2 ft. The exposure rate was 750 frames per second. The lens was operated at $f/2.7$.

A single lens camera equipped with a beam splitter located ahead of the objective was used for the stereoscopic pictures, Fig. 2. The films were projected and viewed using a beam splitter and polaroid system, Fig. 3. These techniques are well established and are described in the literature. (4). The stereoscopic films were taken only in the absolute frame of reference. Since it is necessary to place the axis of the optical system on the axis of rotation when using the rotoscope, binocular or stereoscopic viewing is not possible with this device. Rotation of the camera and associated equipment was considered neither feasible nor warranted at this time.

The tracers, colored white, were a solution of dibutyl phthalate and kerosene proportioned to give a mixture of the specific gravity of the water. The globules formed by this mixture are immiscible in water and, except for surface tension effects which are very small, they may be considered to reproduce accurately the paths of the water. The more common mixture of carbon tetrachloride and benzene acts as a plastic solvent and proved destructive in this particular instance. The tracers were injected through 1/16 in. dia. capillary tubes placed at various points in the stationary and rotating members, depending upon the requirements of the research. Observations were made some distance away from the point of injection, in order to minimize the effect of the tubes and the fluid injection on the flow pattern.

Total Head and Power Determination Techniques

To determine the head generated, total head measurements were made before and after the impeller. The inlet head was determined both directly by a total head tube placed on the centerline in the throat of the inlet nozzle about 7/8 in. below the lowest leading edges of the vanes, Fig. 4, and indirectly by making pressure measurements at the inlet pipe wall some distance upstream from the nozzle just after a honeycomb straightening section. Assuming a uniform velocity profile to exist at this point, the total head was found by calculation.

The question arises as to the reliability of measurements made close to the impeller. It is reasonable to suppose that disturbances caused by the impeller will have little effect on the upstream energy determinations. Thus, although it is not possible to state the exact degree of precision, it can be concluded that these measurements are reliable indications of the true mean specific energy content of the entering fluid. Comparative studies showed that except for a small variation, which averaged about one-half of one per cent and can be partly attributed to losses between the two points, the two heads determined simultaneously tracked consistently down to a unit flow rate of about 0.03. Below that point the variation increased and at shut-off amounted to 9 per cent, the total head tube near the eye indicating high. Detailed studies of the exact cause and nature of the inlet energy deformation at extremely small capacities have not been made. For the present it is sufficient to note that the measurements made with the inlet total head tube close to the impeller gave reliable results over the range of operation herein reported.

For the most part a pitot total head probe was used to determine the specific energy of the fluid at the discharge. The tube was placed at various elevations and distances from the impeller periphery within the collector, Fig. 4. The tube was so mounted that its nose could be placed and maintained at the desired point within the collector and at the same time the tube could be aligned with the mean flow direction in the radial plane as indicated by the thread streamers. Since this type of tube is relatively insensitive to large angles of yaw in all directions, Fig. 5, it was found that this technique produced maximum readings fully consistent with those obtained by the more laborious process of hunting directly for the maximum.

Unfortunately, it is not possible to place the same degree of confidence in the discharge measurements as in those made at the inlet. In certain regions of operation the flow at the periphery becomes extremely irregular and presents almost insurmountable measuring difficulties. Measurements made at a more favorable location some distance from the disturbance are of little significance because of the inevitable losses between such a point and the impeller. To interpret the meaning of the results it is necessary to consider the flow conditions prevalent at each location and flow rate and to evaluate qualitatively the various effects on the tube readings. Such an analysis will be attempted in a later section of this report. Let it suffice to say at this point that the curves presented should not be accepted per se as indicating the true head-capacity characteristics at all flow rates.

Since it proved convenient to operate the impellers at relatively low speeds, it was necessary that the vertical dynamometer be equipped to measure torques of small magnitudes. The dynamometer was cradled on a large angular-contact ball-bearing which carried the full dynamometer and impeller load. Suspension of the assembly to improve response was found unnecessary; the slight inherent vibration in the machine adequately overcame any tendency towards sticking. The torque was picked off with an unbonded strain gage which formed two legs of a Wheatstone bridge. A null type circuit was used incorporating a helipot as the variable resistance and an optical galvanometer as the null point indicator. This arrangement proved very successful.

By power input is meant that power imparted to the fluid as it passes through the impeller. To eliminate the viscous pumping on the outside surfaces of the shrouds, the water on the top shroud was eliminated and the viscous drag on the periphery and suction shroud was tared out before water was admitted to the impeller passages.

Range and Accuracy of Data

The rotative speeds of the impellers ranged from 150 to 360 rpm.

The magnitude of the differential head readings in some cases exceeded four feet. At the most prevalent test speed of 180 rpm, in the region of the best efficiency point, they averaged about 1.8 ft, and could be read with an accuracy of ± 0.2 of one per cent.

Capacities ran up to 1.75 cfs. Again, at 180 rpm they averaged about 0.5 cfs. These flow rates were determined by calibrated venturi meters to an accuracy of $\pm 1/2$ of one per cent.

Input power ranged up to 0.6 hp. It averaged at 180 rpm, 0.125, and could be determined at that speed with an accuracy of $\pm 1\%$.

Results of Total Head and Power Measurements

The results of tests made on the four impellers are presented in Figs. 6 to 11. Figs. 6 to 9 contain the several so-called head-capacity curves for each impeller which were obtained with the discharge pitot tube located at the various stations indicated in Fig. 4. Figures 10 and 11 show the brake horsepower, the average head-capacity and efficiency curves for Impellers III and IV, and the in-case performance curves of these impellers. Also presented in all six figures are the ideal head-capacity curves based on Busemann's solution of a two-dimensional radial-flow impeller with logarithmic spiral vanes (5).

The curves are all presented as dimensionless unit plots to facilitate comparison. Unit head is defined as

$$\frac{Hg}{u_2^2}$$

where u_2 is the impeller tip speed in fps and H is the total head generated in feet. Unit capacity is defined as

$$\frac{C_{m2}}{u_2}$$

C_{m2} being the mean radial or meridional component of the fluid velocity at discharge in fps and equals the flow rate through the impeller divided by the net peripheral discharge area. Unit brake horsepower is defined as

$$\frac{BHP}{u_2^3}$$

Results of Flow Observations

The results of the flow observations will not be presented under a separate heading but will be introduced when considered significant in the analysis of the results presented in Figs. 6 to 11.

Significance of Total Head Measurements

In pitot tube surveys of this type, there are several basic causes of error. Poorly constructed tubes of unknown characteristics and calibration can introduce errors. The tubes used were all carefully constructed and their characteristics and

coefficients determined by experiment, Fig. 5. Poor response of the entire pressure measuring system may cause inaccuracies. The pitot tubes and the associated manometer system formed a highly damped system and thus, assuming variations only in magnitude of the approach velocity, faithfully indicated the mean total head. Excessive angles of yaw or, as in the case of cylindrical tubes, the angle of pitch, can introduce the most serious error. This error can arise from two sources: first, although as noted above, variations in magnitude of the approach velocity will not affect the results, angular variations of the turbulent component can introduce a negative error in the total head indications even though the tube may be aligned with the mean flow direction. The high degree of turbulence arising in the impeller and present in the discharge flow has been reported previously (1). Qualitative tracer studies indicated the mean temporal velocity variations to be about ± 35 per cent of the mean through-flow velocity. This corresponds to an angular variation of the approach velocity of approximately ± 20 degrees. From Fig. 5 it is seen that this variation approaches the maximum that the total head tube can handle satisfactorily. The presence of the velocity fluctuations undoubtedly affected the measurements; however, it is believed this error was small. The second source of error due to yaw arises from direct misalignment of the tube with the mean approach velocity. Since, from practical considerations the tubes used in this study were set assuming horizontal flow parallel to the shrouds, and could be set in the radial plane to coincide only with the mean of the approach velocity, errors from this source could be large and the deciding factor as to the significance of the results. To consider this point further, it will be necessary to analyze the discharge flow patterns at the tube stations at various flow rates and by both direct and indirect means attempt to estimate the error present.

To simplify the following discussion, the differential total head measured with the discharge pitot tube at Station A, Fig. 4, will be referred to merely as head A or H_a , when at Station B, as head B or H_b , etc. Reference also will be made to Fig. 12, which presents plan and elevation views of various discharge flow patterns. Although the figure refers specifically to Impeller IV, the drawings were synthesized from a number of photographic and visual observations on all four impellers and may be considered typical of the series. The vectors shown are all equal in length and indicate mean flow directions only. They are drawn in true orthographic projection.

The four sets of head-capacity curves, Figs. 6 to 9, while differing in some details, all show the same general characteristics. Beginning at zero head, high capacity, the curves in each figure rise almost linearly. Beginning with little spread they diverge gradually at first, then rapidly. Suddenly, the curves jump together and continue in this pattern to shut-off.

Consider that portion of the curves, H_a , H_b , and H_c , from maximum unit capacity to the point where they start to break away from each other rapidly. In this zone of operation it was found that the absolute discharge angle α_2 was always positive and, although not necessarily uniform over the entire breadth, was relatively constant at any one elevation. Since the α_2 distribution was not completely uniform, cross flows normal to the shrouds were present. These flows were observed as a roll or twist in the

flow and were found to exist in the impeller passages and in the collector. Obviously, the greater the angular difference and the greater its extent over the impeller breadth, the stronger will be the cross-flows. At high capacities, Fig. 12(a), α_2 was large and fairly uniform across the breadth except near the bottom shroud. It was found that in this operating region the cross-flows were small and being localized near the lower shroud did not affect greatly the bulk of the flow. In the region of the best efficiency point, Fig. 12(b), α_2 varied almost uniformly across the breadth, being least near the bottom and greatest near the top shroud. However, the total angular difference was only about 10° , and the cross-flows that developed tended to deflect the discharge only slightly towards the top shroud, as can be seen in the figure. It is concluded that, since α_2 was relatively steady and strong cross-flows were not present, errors introduced due to yaw were small in this region and that considerable confidence can be placed in the results as indicating to a fair degree of accuracy the actual mean specific energy at the various stations.

It is significant to note that, particularly in the case of Impellers III and IV, the H_a , H_b , and H_c curves show very little divergence. Surveys conducted up to within $1/8$ in. of the shrouds showed this same energy uniformity. This means that, as a first approximation, to obtain the mean head-capacity curves of these impellers in this zone of operation, it was not necessary to perform the laborious task of weighting all head readings with an appropriate flow rate and averaging, but rather it was possible to take directly a linear average of the heads and not introduce serious error. This fortuitous circumstance allowed calculation of water horsepower, and, hence, knowing input power, the impeller efficiency.

Upon comparison with the model in-case efficiencies, Figs. 10 and 11, note that the impeller efficiencies fall as they should between the former and 100 per cent. At the design point the model losses were about 11 per cent. Since we know losses are distributed between the impeller and the model case, it is probable that the maximum error in the head curves at the design point is less than $\frac{1}{3}$ per cent. Since no noticeable changes occurred in the discharge flow pattern at capacities above the design point, it can be assumed the general order of magnitude of the error will remain the same throughout the region of operation.

As a further check on the tube indications in this region, surveys were made in the weir opening where an accelerated flow existed. Since the peripheral flow pattern was found fairly uniform at these capacities, the losses between the two stations should have been small and the total heads in fair agreement. Fig. 13 shows the results of this survey compared with the mean total head.

Consideration will now be given to the region in which the head curves, H_a , H_b , and H_c , show a relatively wide divergence. As the flow rate was reduced below the design point, not only did the mean α_2 decrease but the difference between α_2 at the top and bottom shrouds increased, indicating a non-uniform meridional or through-flow velocity profile, this velocity being least near the bottom and greatest near the top shroud. Not far below the best efficiency point α_2 near the suction shroud, followed progressively

by α_2 at stations C, B and A, approached zero as the flow rate was lowered, Fig. 12 (c) and (d). It was observed that in this region of operation strong cross flows, which appeared as a spiralling toroidal roll developed in the collector. When α_2 first approached zero, the cross flows extended in to the impeller periphery and swept across the lower portion of the impeller breadth. At lower capacities the effect spread over a greater percentage of the breadth. Photographic studies showed that particles in the impeller near the suction shroud were picked up by this roll and were carried spirally upward, and left the passage near the top of the discharge area. It is believed that this transverse component of the flow was of sufficient magnitude to cause the absolute flow velocity to approach the tube locations at a yaw angle too great for satisfactory tracking of the tube and caused the total head indications to be erroneously low, Fig. 12(c) and (d).

Experimental evidence seems to substantiate the above discussion. A cylindrical type total head tube, with the hole drilled normal to its axis, was inserted in the flow at station A, Impeller IV. Such a tube is much more sensitive to pitch in the plane of its axis, which here corresponded with the impeller axis, than the pitot probe, Fig. 5. If the argument were correct it would be expected, assuming the cross-flows to develop gradually, that measurements made with this tube would break away from those made with the pitot probe at the same station at a higher flow rate. Fig. 14 presents the comparison which is favorable to the analysis. If these transverse flows progressed over the impeller breadth as the capacity was lowered, tube stations B₃ and B₂ should have been affected first and in this order. Stations C, B, and A should have followed in like manner. Fig. 6 shows this to be the case. A tube was placed 1/8 in. below the top shroud of Impeller IV and its results compared with those of station A. The former curve showed a continual rise to a lower flow rate than any of the other tube readings. Note also, Fig. 13, that total head measurements made at the weir opening actually exceeded those made at stations B, C, and B₂, thus further substantiating the contention that the total head readings in this region are erroneously low. An analysis of the efficiency curves in this region of operation yields little fruit. The losses were of such magnitude as to make a reasonable estimate of the error in the head-capacity curves impossible.

Consider now the remaining portion of the head curves. When α_2 near the top shroud approached zero an abrupt change took place in the discharge pattern. Over the entire breadth α_2 fluctuated positive and negative, indicating that the flow reentered the impeller passages in some areas. Photographic observations showed reentrance to occur in the region near the leading face of a vane. The heaviest net through-flow occurred in the central portion of the discharge breadth. Flow conditions in this zone were unstable. The unit capacity at which this pattern change occurred depended upon whether it was approached from the high or low capacity side. Asymmetries in the discharge weir or within the collector, such as the number of pitot tubes, produced visible effects on the discharge flow pattern. The discharge from between the collector weirs was characterized by a pronounced pulsing. The single transverse flow noted previously disappeared and two opposing flows developed, Fig. 12(e).

Since all stations were subjected to approximately the same flow conditions, it is to be expected that all head measurements would more or less agree. The question is, what is their significance. In a precise energy survey it would be necessary to measure, in the relative plane, the head at each point about the impeller, weigh each reading with its proper differential flow rate, paying due attention to sign, sum up the results and divide by the total net discharge. In this summation all energy determinations along reentrant stream lines would appear as negative quantities. When a stationary tube is used to integrate and give an average head reading, the results will be erroneously high. Re-entrance occurred at low discharge when the absolute velocity left and reentered the impeller at a small angle to the tangent. Since the total head tube used was relatively insensitive to yaw, it was in no way able to differentiate between the additive and subtractive energy pulses and hence the integrated resultant head measurement can be expected to be high. Whether this effect was great enough to account entirely for the pronounced jump in the head curves is questionable. It is believed that in this region a major shift took place in the flow pattern throughout the impeller and that this shift may have so affected the process of energy transfer as to give rise to a sudden change in the total head generated. Although dominant reentrance was not the only phenomenon affecting the measurements. Eddies and whirls causing cross-flow were present in varying and indeterminant extent. Since both the in-case and free impeller losses were great in this region, little help could be gotten from the efficiency curves. The losses in the collector being of unknown magnitude, measurements at the weir were of little significance. At flow rates below the point of convergence of the head curves, it is not only impossible to state the accuracy of the measurement but it is highly questionable if even the general trend shown is correct.

The curves were not continued below a unit capacity of 0.03 because of inherent inaccuracies in the discharge and inlet head measurements.

Impeller Characteristics and the Busemann Line

It is of interest to note the close correlation between the head-capacity curve of each impeller and the corresponding ideal curve computed on the basis of Busemann's potential flow analysis of a two-dimensional impeller with logarithmic spiral vanes (5). In the region of minimum loss, the total head generated exceeded the Busemann value, Figs. 10 and 11. Experimental error will not explain the difference. Although their relative significance is not known, it can be said this discrepancy lies in real fluid effects, the deviation of the vane curvature from a log-spiral, and the three-dimensional aspect of the test impellers. The fact that the trend of the impeller curves deviate from the Busemann lines, either side of the best efficiency point may be attributed in part to losses.

Effect of Design Variations on the Head-Capacity Curves

In Fig. 15 are shown the mean head-capacity curves for all four impellers listed. The approximate location of the design point of the impeller is marked on the curves.

Inlet vane angle. Note that the head-capacity curves of Impellers I and II drop off more rapidly at high capacities than do the Impeller III and IV curves. It is believed that the smaller inlet vane angle combined with the other slight alterations in the impeller eye were sufficient to give rise to large losses in the transparent impellers at high flow rates and thus cause the variation. This contention has been substantiated in part by the observation that at zero head the flow discharging from Impellers I and II had a much greater mean positive tangential component than did the discharge from Impellers III and IV.

In the case of ideal, frictionless flow, when the total head generated is zero, it is possible to have a positive whirl component C_{u2} of the absolute velocity at discharge only if a positive prerotation exists in the flow entering the impeller. In a real viscous fluid medium it is possible to obtain this same situation without prerotation. In this case the specific energy lost, H_L , would be represented by
$$H_L = \frac{C_{u2}u_2}{g}$$

Large scale or mass prerotation in the inlet of sufficient magnitude to account for the angular momentum noted at the discharge, has never been observed in these tests of free impellers. Small, local perturbations at the leading edges of the vanes have been found but most of the flow entering the channels has had no tangential velocity component. Hence any rotation in the entering flow is of a small order of magnitude, except at extremely low flow rates near shut-off. This does not infer that large-scale prerotation does not exist in a complete pumping unit. Asymmetries present in discharge volutes impose quite a different situation on the impeller. These asymmetries may give rise to reverse flows in the impeller of sufficient magnitude to extend back into the inlet and thereby supply the mechanism necessary to start and sustain prerotation. The observation does imply that prerotation is not caused by the viscous drag of the impeller and its fluid on the incoming flow and also that the fluid does not follow the path of least resistance and hence automatically rotates just the right amount to assure smooth entrance conditions.

It remains to be shown that losses of sufficient magnitude were present within the impeller passages. At this operating point a large zone of separation was observed along the leading or convex surface of the vane near the inlet. That this zone must have been present follows directly from the previous statement that prerotation was small and thus the incoming flow did not match the vane setting at off design point operation. In this region, back flows which extended down to the vane inlet edge and beyond were observed. Although it is not possible to state definitely at this time, it is believed that these flows did not extend far into the eye. The momentum transferred to the entering fluid in this manner assisted in maintaining the small prerotation previously discussed. It is reasonable to assume that such a zone

of separation would give rise to losses. The degree of the separation, and thus the losses at a fixed flow rate, will increase as the inlet angle decreases and the mismatch becomes greater; hence the more rapid drop in the head curves of Impellers I and II. The order of magnitude of these losses for Impellers III and IV is represented in Figs. 10 and 11 by the brake-horsepower at the zero head flow rates. Quantitative data as to the magnitude of C_{u2} are not pre-rotation, C_{u2} and α_2 were calculated from the brake horsepower data. The angle α_2 was found to be of the same general order of magnitude as the mean angle indicated by thread streamers, thus adding credibility to the argument.

Discharge vane angle. The effect of varying the discharge vane tip angle is to change the slope of the head-capacity curve. This is evident in Fig. 15 upon comparison of Impeller III which had a discharge angle of 17° and Impeller IV with an angle of $23\frac{1}{2}^\circ$. Figs. 6 to 11 show the close agreement between the actual and the ideal or theoretical slopes in the region of the design point.

Number of vanes. The effect of the number of vanes on the head generated is shown in Fig. 15. Again, referring to Figs. 6 to 11, Impellers I, II and IV which all had approximately the same discharge angle, note the close correspondence over the range near the design points of the actual trend with the theoretical.

Cylindrical vs skewed discharge vane tips. It will be noted upon comparison of Figs. 6 to 9, that the head curves of Impellers III and IV show less spread at high capacities than the curves of Impellers I and II. As the curves slowly diverge, Impellers III and IV show higher specific energy fluid near the suction shroud, whereas the transparent impellers show the reverse situation. It is believed that the difference in the discharge vane tips may have been the primary factor causing these variations. However, the inlet differences previously mentioned also may have been significant and hence a statement as to the effect of slewing the tips only cannot be made.

The importance of the above discussion does not lie in the effects of the various design parameters on the impeller characteristics at the design point. The results, deducible from elementary theory, have been known for years and are used daily in pump design. The real significance is to be found in the fact that the head-capacity curves of the free impellers tested followed closely the trends indicated by two-dimensional theory over a relatively wide range of operating conditions. This fact stands in stark contrast to the customary deviations between this theory and the head curves of most complete pumping units and accentuates the profound influence of the volute or diffuser in determining the over all operating characteristics of a centrifugal pump.

Comparison with In-Case Characteristics

Figures 10 and 11 show the characteristics of impellers III and IV with the corresponding in-case or complete pump performance. At the best efficiency points considering the estimated accuracy of the free impeller head curves, it is possible that the entire discrepancy between these and the in-case curves be charged to volute and leakage losses. However, this is not

believed to be entirely justified. Note that the unit brake horsepower curves of the model pumps which include mechanical losses and disc friction, which in this case was a loss since the models were diffuser type pumps, reduce to almost exact correspondence with the free impeller results which exclude these losses. It would be expected that the latter curves would fall below the model results by the amount of these losses, which as a rough estimate, might run about 3 per cent of the total input horsepower. That this is not the case indicates the failure of the similarity laws and suggests a Reynolds number effect and/or a fundamental difference in the flow through the in-case impeller as a result of the presence of the volute. Input power tests at higher speeds have shown a small Reynolds number effect. With the present techniques, head measurements could not be made with sufficient precision to detect such an effect in the head generated. Direct studies of the influence of the volute or diffuser on the impeller flow pattern have not been made. However, from a consideration of the total available losses in the in-case units it is felt that such an effect is negligible. In the region of the design point the discrepancy between the in-case and free impeller head curves can be charged in part to a small Reynolds number effect and the large remainder to losses within the diffuser.

This is not believed to be the case at off design points. The presence of prerotation in a complete pump at off design points and its almost total absence in a free impeller at any operating point indicate the strong influence of the volute or diffuser on the flow through the impeller. It is believed that the discrepancies in the brake horsepower curves at off design points are due to the diffuser. Likewise a greater part of the deviation between the head curves may be attributed to the change in flow pattern than was true at the design point. However, it is believed that the greatest percentage of the difference arises from increased losses both within the impeller and within the volute or diffuser. This is shown in the efficiency curves which for the free impellers are much flatter and indicate the impellers themselves to be capable of such wider ranges of operating conditions at high efficiencies than the in-case models.

The above discussion emphasizes the important part played by the volute in determining the magnitude of the losses at the design point and the form of the brake horsepower head and efficiency curves at off design points.

It is significant to note that the zones of instability are present in both the in-case and the free impeller results. This being the case it is strongly suggested that the instability is caused by a feature of the impeller design. It has been reported (2) that the inlet vane angle is the determining factor. In the same report it is stated that the diffuser design governs the magnitude of the break in the head curve. Unfortunately, due to the curvature of the blades at inlet, detailed studies in this zone were not possible and hence the work so far permits no amplification of the first point. A tentative explanation of the second point is that in the region of the instability the highly irregular discharge flow pattern gives rise to losses. The degree of these losses and for that matter, the very character of the flow pattern since it is extremely unstable will most certainly depend upon the volute design.

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ILLUSTRATIONS

<u>Figure</u>	<u>Title</u>
1	High-speed motion picture camera equipped with rotoscope.
2	High-speed motion picture camera equipped with beam splitter for stereoscopic photography.
3	Motion picture projector equipped with polarizing beam splitter for stereoscopic projection.
4	Total head tube stations.
5	Sensitivity of total head tubes to change of direction of approach velocity.
6	Head-capacity characteristics of Impeller I.
7	Head-capacity characteristics of Impeller II.
8	Head-capacity characteristics of Impeller III.
9	Head-capacity characteristics of Impeller IV.
10	Complete characteristics of Impeller III.
11	Complete characteristics of Impeller IV.
12	Approximate flow patterns at discharge of Impeller IV.
13	Mean total head measurements at impeller periphery and at collector weir, Impeller IV.
14	Effect of total head tube characteristics on head-capacity curve, Impeller IV.
15	Mean head-capacity characteristics of four test impellers.

	Outlet Dam in.	Outlet Width in.	Eye Diam. in.	Outlet Vane Angle	Number of Vanes	Designation
Transparent Impellers	10.3	1 1/4	5 1/2	24	5	I
	10.3	1 1/4	5 1/2	24	6	II
Metal Impellers	12 7/8	1 9/16	6 7/8	17	5	III
	12 7/8	1 9/16	6 7/8	23 1/2	8	IV

Table I - Principle Impeller Dimensions.

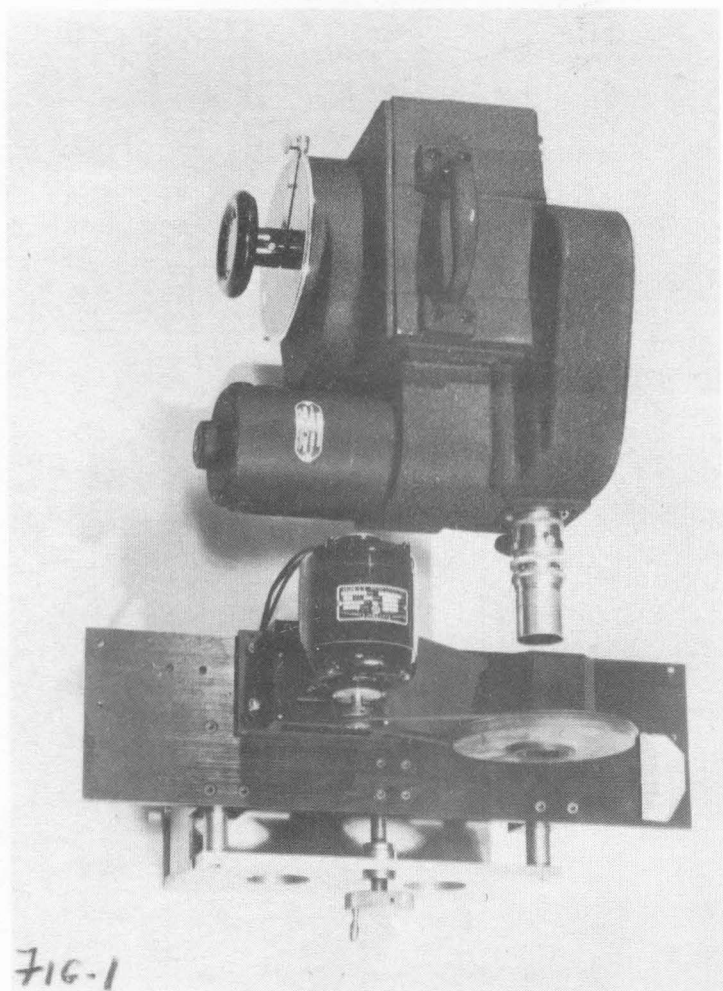


FIG-1

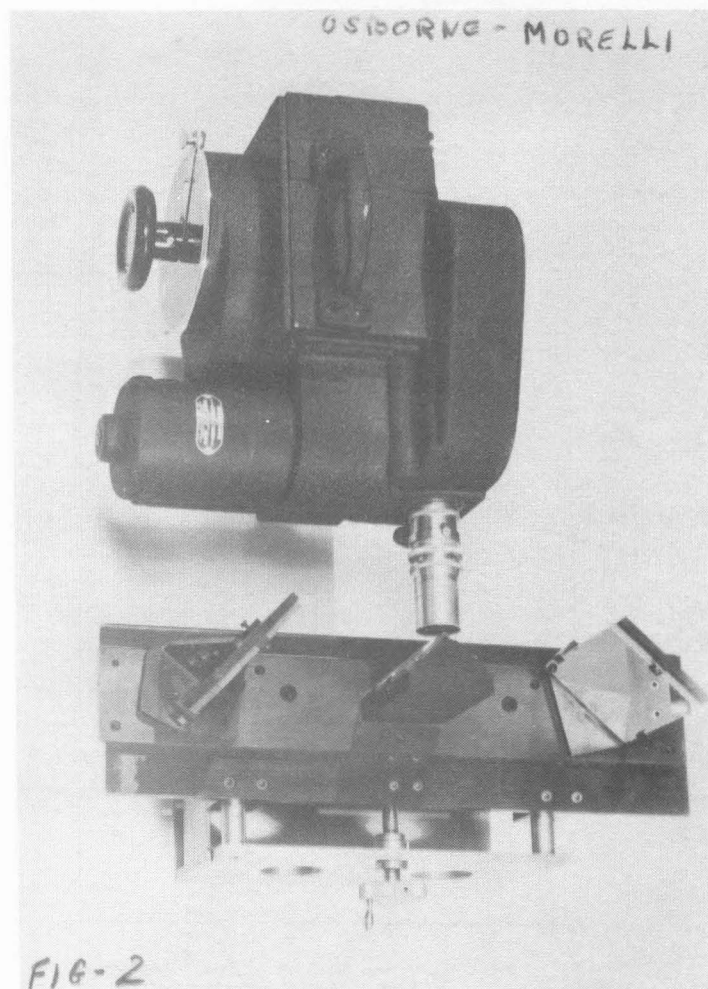


FIG-2

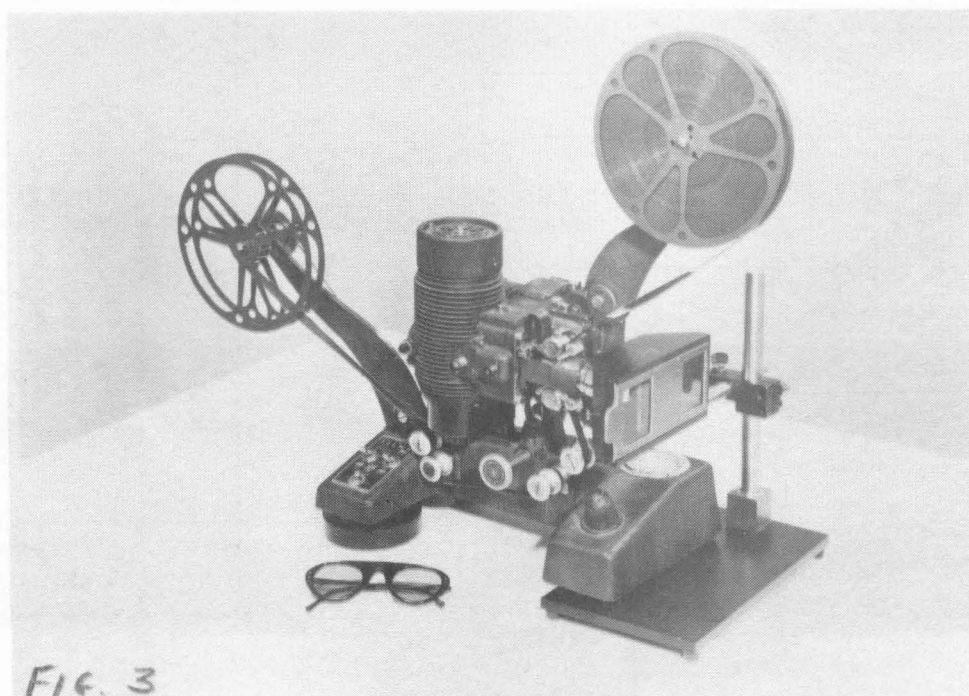


FIG. 3

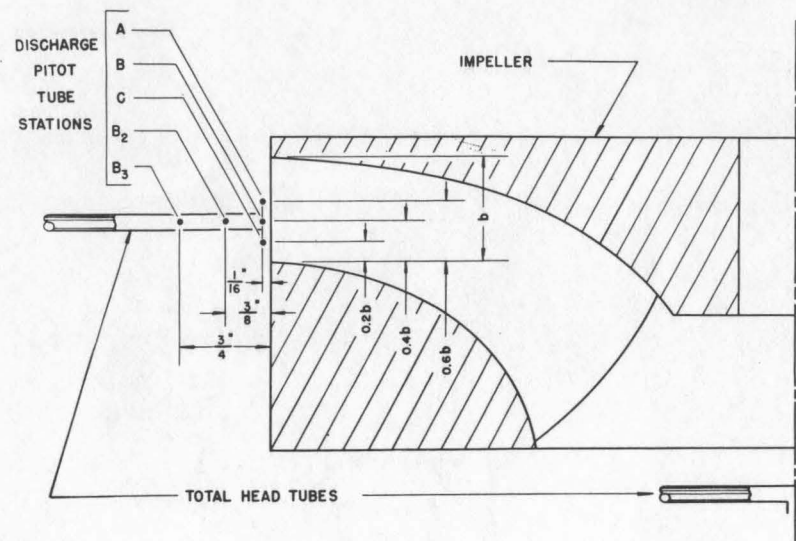


FIG-4

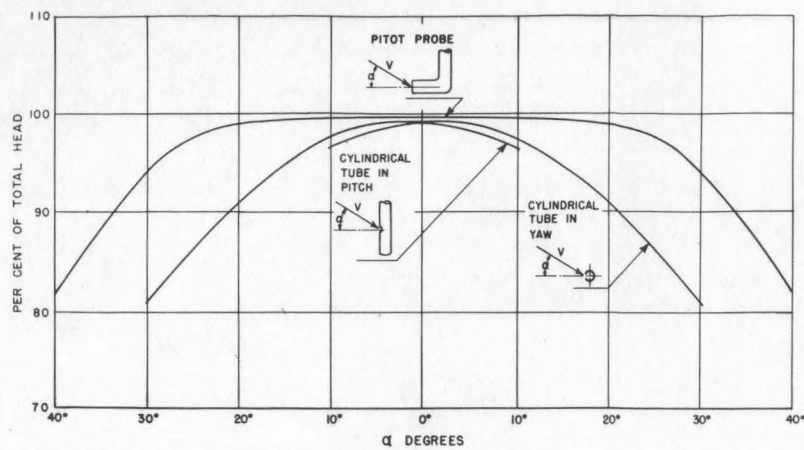


FIG-5

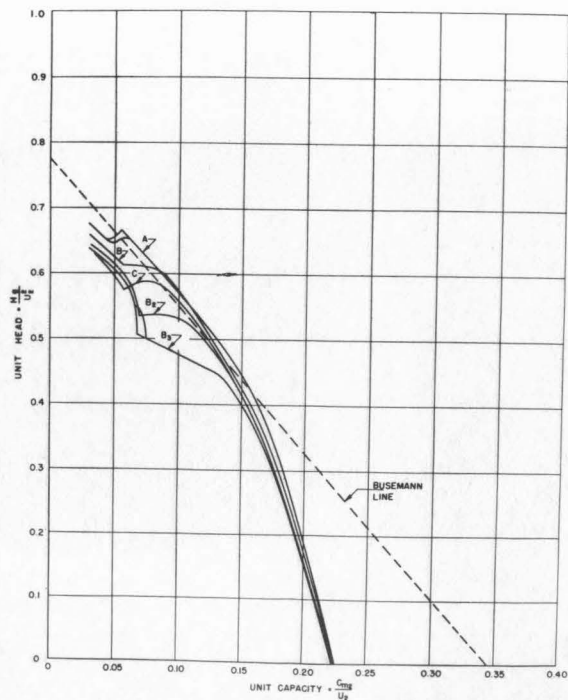


FIG-6

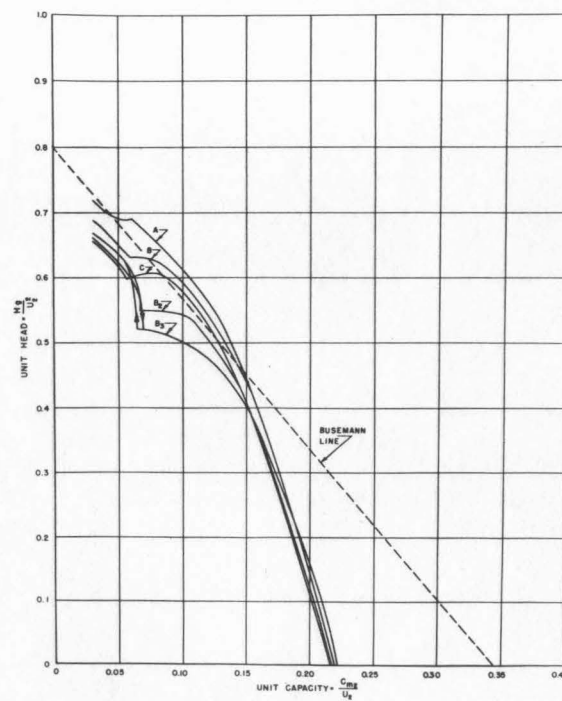
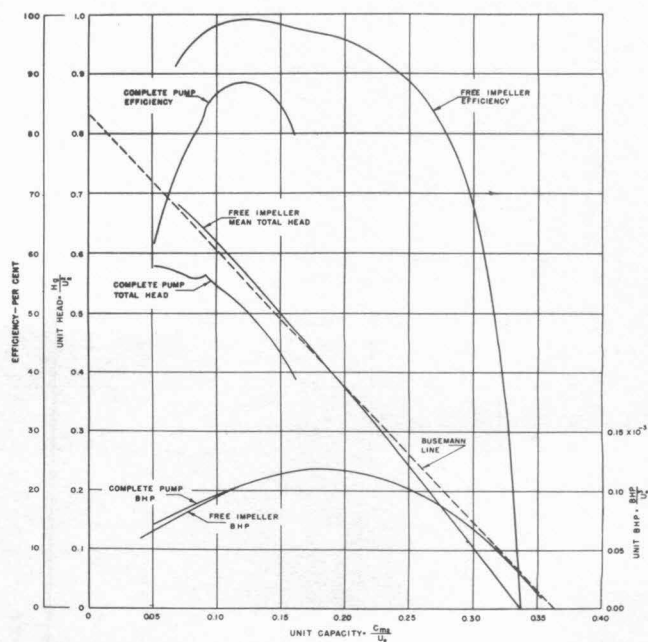
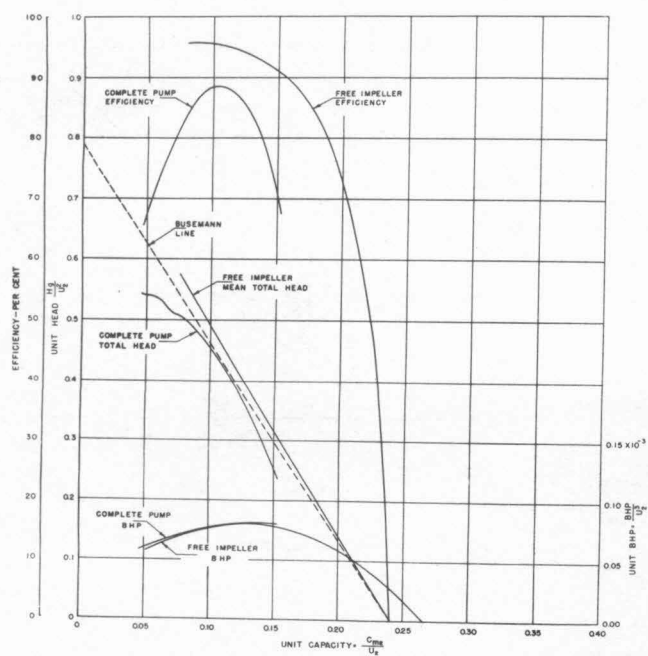
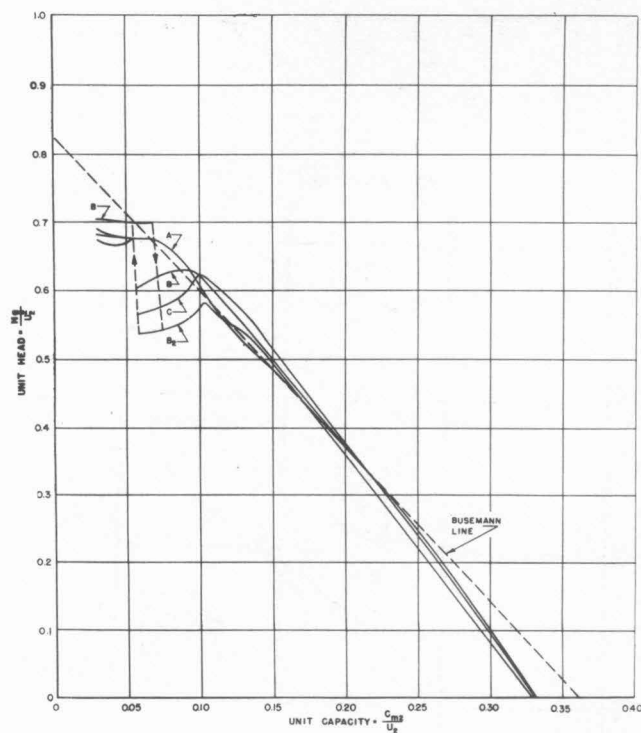
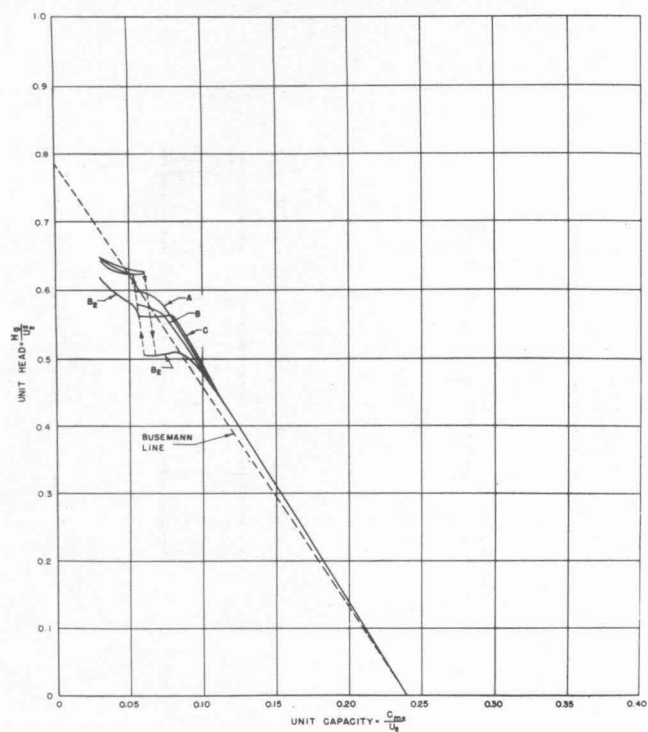


FIG. 7



OSBORNE

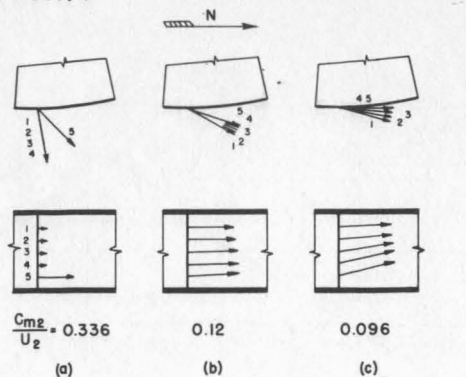


FIG-12

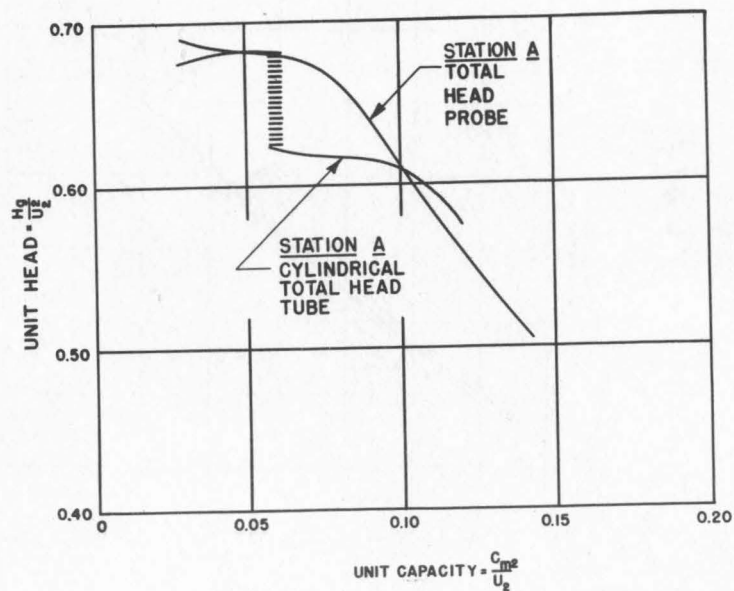


FIG-14

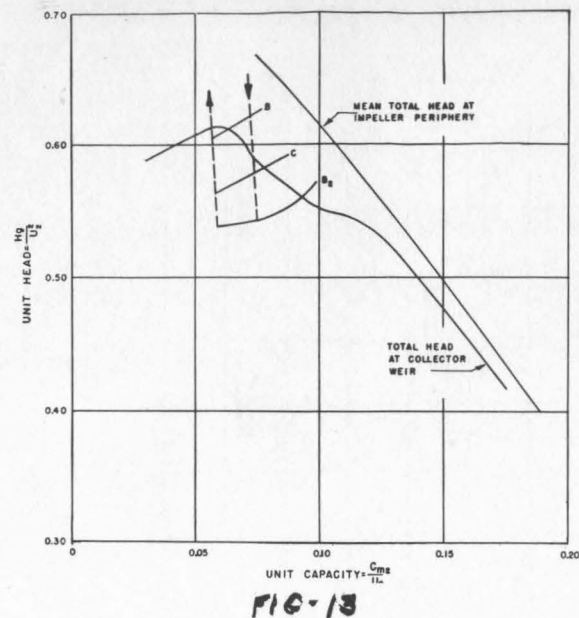


FIG-13

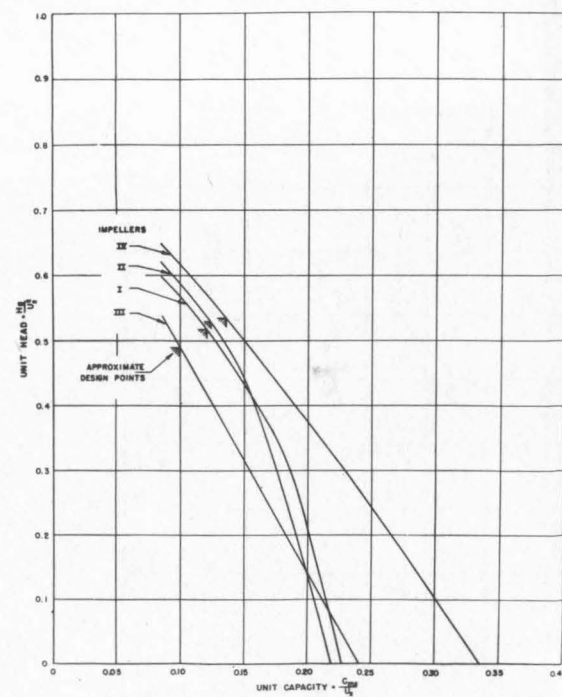


FIG. 15